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TECHNICAL REPORT ARLCB-TR-83040

FATIGUE LIFE ANALYSIS AND TENSILE OVERLOAD EFFECTS WITH HIGH STRENGTH STEEL NOTCHED SPECIMENS

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NOVEMBER 1983

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REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER ARLCB-TR-83040	2. GOVT ACCESSION NO. AD-A136065	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) FATIGUE LIFE ANALYSIS AND TENSILE OVERLOAD EFFECTS WITH HIGH STRENGTH STEEL NOTCHED SPECIMENS		5. TYPE OF REPORT & PERIOD COVERED Final 6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) J. H. Underwood		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS US Army Armament Research & Development Center Benet Weapons Laboratory, DRSMC-LCB-TL Watervliet, NY 12189		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS AMCMS No. 6111.01.91A0.0 PRON No. 1A325D021A1A
11. CONTROLLING OFFICE NAME AND ADDRESS US Army Armament Research & Development Center Large Caliber Weapon Systems Laboratory Dover, NJ 07801		12. REPORT DATE November 1983 13. NUMBER OF PAGES 18
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASSIFIED 15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

Presented at IX AIRAPT International High Pressure Conference, Albany, NY 27-28 July 1983.
Published in proceedings of the conference.

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Fatigue Life
High Strength Steel
Residual Stress
Notched Bend Test
Stress Concentration

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Fatigue crack growth results are presented for a series of tests of high strength steel notched-bending specimens. Eight values of stress concentration factor from 1.5 to 4 were represented in the tests, as well as six forging procedures with yield strengths from about 1000 to 1200 MPa. The cyclic lives of the specimens, ranging from about 2000 to 100,000 cycles, were analyzed, using fatigue stress range calculated from stress concentration factor and from

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20. ABSTRACT (CONT'D)

a fracture mechanics method. A statistical comparison of the two methods was performed. Photoelastic and finite element methods were used to obtain some of the notch root stresses.

The effects of a single prior tensile overload on fatigue life were considered for many of the tests. There appeared to be a critical ratio (about unity) of cyclic stress range at the notch root relative to yield strength, below which a tensile overload extended fatigue life and above which a tensile overload shortened life.

The effects of prior thermal overload on fatigue life were also investigated in five tests of two specimen geometries. Rapid cooling of the outer diameter of a hollow disk segment of a cylinder extended the fatigue life in subsequent cyclic bending testing of the segment.

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OBJECTIVE

Pressure vessels often have notches or other stress concentrations present. Considering further that pressure vessels are nearly always subjected to some cyclic loading, fatigue cracking at notches is an important problem. The objective here is to describe some fatigue life testing and analysis which was performed with notched specimens in order to determine the effects of notch overload on fatigue life of pressure vessels.

The overloading of a notched component is parallel in many aspects to the overstraining of a thick-wall tube. In the same way that compressive circumferential residual stress produced near the inner diameter of an overstrained tube increases the fatigue life (ref 1), the tensile overload of a notch can increase notch fatigue life (ref 2). The increase in life of an overloaded notch can be attributed, as with an overstrained tube, to compressive residual stress, in this case produced locally at the notch root. The tests and analysis described here identify some of the conditions and effects of notch overload.

TEST PROCEDURES

Specimens

Previously, internal pressure fatigue tests of pressure vessel sections were performed (ref 3), but they took considerable time. In order to investigate a variety of test conditions, simpler laboratory tests were used (refs 2,4). Figure 1 shows the two general types of lab tests used in previous work (refs 2,4) and in the results here. Details of the test

References are listed at the end of this report.

specimens will be given in the upcoming discussion of Table 1. The hollow disk specimen shown as Figure 1(a) is sliced out of a relatively thick-wall cylinder, that is, one with outer to inner diameter ratio of about 2.0. It is loaded in compression, so that the area of the notch is subjected to tensile circumferential stress. The tensile-loaded specimen, Figure 1(b), is similar to the compact specimen used for fracture testing (ref 5). A half-disk specimen, called the arc specimen (ref 5) was also used in a similar way to that shown in Figure 1(b), for forging #2 tests.

Test Conditions

Eight different notches were tested with stress concentration factors, K_t, from about 1.5 to 4. See details in Table 1, which also shows the notch root radius, ρ, and other pertinent dimensions. The material for all tests was ASTM A723, Grade 2, a high strength nickel-chromium-molybdenum steel used for pressure component forgings. The test material was taken from six forgings with somewhat different manufacturing processes, but all included a vacuum degassing process. The yield strength, σ_y, ranged from 1030 to 1200 MPa, as listed in Table I.

The loading of the test specimens was constant amplitude fatigue at from 3 to 30 Hz, with maximum load, P_{max}, and load ratio, $R = P_{min}/P_{max}$, as shown in Table I. The general level of load was chosen so that fatigue lives would be in the relatively low-cycle range, that is, from about 1000 to 100,000 cycles. A crack of a few millimeters surface length typically developed on the notch root surface at about one-half of the eventual fatigue life, which was determined when the specimen broke in half. The average fatigue lives, $\bar{N}$, listed in Table I, are the mean of two replicate tests, except for the three single tests indicated.

TABLE I. CONDITIONS FOR LIFE AND OVERLOAD TESTS

Specimen Geometry							Material		Loading		
K_t	ρ mm	a mm	h mm	W mm	B mm	x mm	Forge #	σ_y MPa	P_{max} kN	R	$\bar{N}$ Cycles
2.18	3.4	16.0	6.8	40.0	10.0	28.0	1	1040	+15.6	0.10	83,000
1.53	12.7	12.7	25.4	43.7	25.4	136	2	1030	+25.4	0.10	29,800
1.63	18.0	50.8	36.0	102	25.4	76.0	3	1110	+111	0.10	18,700
3.80	1.3	11.6	15.8	59.5	50.8	140	4	1190	-190 -76.0	0.42 0.42	4,330 74,400
1.98	7.9	11.6	15.8	59.5	50.8	140	4	1190	-190	0.42	10,200
1.95	6.4	25.4	12.7	57.2	25.4	126	5	1200	-34.5 -23.6 -15.6	0.10 0.10 0.10	4,900 (1) 16,200 (1) 44,600
2.41	1.5	5.0	25.4	57.5	50.8	194	6	1170	-175 -132	0.54 0.39	9,800 41,700
3.17	1.5	10.0	25.4	57.5	50.8	192	6	1170	-175 -132 -52.9	0.54 0.39 0.39	2,780 4,170 82,200 (1)

RESULTS AND ANALYSIS

K_t Approach

Results of twenty-five notch fatigue tests are shown in Figure 2(a). Fatigue life is plotted versus maximum stress, σ_{max}, normalized by σ_y. As shown, σ_{max} is calculated as the product $\sigma_{nom}\ K_t$. The maximum nominal stress in all tests here is defined as

$$\sigma_{nom} = \frac{P_{max}}{B(W-a)} \left(6 \frac{x}{W-a} \pm 1\right) \quad (1)$$

where B, W, a, x are as shown in Figure 1. Equation (1) describes the sum of bending stress, $6x/W-a$, and either tensile or compressive direct stress, the $\pm$ 1 term, on the unnotched ligament of the specimen. The K_t values used, listed in Table I, were determined from photoelastic analysis (ref 6) for the forging #1,2 tests, from a K_t compendium (ref 7) for forging #3,4,5 tests, and from finite element analysis (ref 4) for the forging #6 tests.

The test results fall into two groups indicated by the dashed lines. Load ratio is the separating parameter, with higher R leading to higher fatigue life, as would be expected. When R is included in the stress parameter in Figure 2(b), so as to change σ_{max} to stress range $\Delta\sigma$, all the test results can be represented by the single least squares line shown. This result is consistent with observations that low-cycle fatigue life is determined by total strain range. Even though it was elastic stress range which was controlled in the tests, this in effect controls the total strain range at the notch root (ref 8). Particularly for high strength notched components, the elastic stresses throughout the component impose a certain total strain range at the notch.

$K/\rho^{1/2}$ Approach

Rolfe and Barsom (ref 8) proposed that the well-known expression for the maximum tangential stress at an ideal elliptical notch can be used to calculate notch root stresses at many notches of practical concern. The expression,

$$\sigma_{max} = 1.12\ K/\rho^{1/2} \quad (2)$$

where K is stress intensity factor, and ρ is notch root radius, is exact only for $\rho \rightarrow 0$, but it provides useful approximations for finite ρ. Figure 3 shows the fatigue life results plotted using Eq. (2) to calculate $\Delta\sigma$. The $(1 + \rho/\rho_o)$ term, with an arbitrary $\rho_o = 10$ mm, was added here so that the data over the whole range of ρ could be represented by one expression. The result is a least squares fit line with very similar slope and correlation coefficient as those using the K_t approach.

Reviewing Figures 2(b) and 3, it is clear that both approaches give an adequate description of fatigue life over a significant range of material properties and notch geometries. These fatigue life descriptions are useful for design as well as for baseline data for investigation of additional effects on fatigue life, such as the effect of mechanical overload, discussed below.

Mechanical Overload

Fifteen overload tests were performed which were identical to the tests summarized in Figure 1 and Table I, except that a single mechanical overload was applied to the specimen before fatigue loading. The overload was in the same direction as the subsequent fatigue loading, and with magnitude such that tensile tangential stress was produced well into the plastic range. The intent was that the elastic recovery upon removal of the overload would leave the notch root with compressive tangential residual stress. The fatigue life of the overloaded specimens was determined and normalized by the mean life of tests with no overload; see Figure 4. Various overload ratios are shown, from $P_{ov}/P_{max} = 1.5$ to 3.0. An increase in overload ratio increases life ratio in some cases, as indicated, for example, by the tests at $\Delta\sigma/\sigma_y = 0.6$. Two tests with P_{ov}/P_{max} = 2.0 gave $N_{ov}/\bar{N}$ of about 2.9, while the test with P_{ov}/P_{max} = 3.0 gave $N_{ov}/\bar{N}$ of 11.2, at the point when the test was interrupted at 500,000 cycles with no crack.

The most significant result of the overload tests was the clear boundary between beneficial and deleterious effects of overload on fatigue life; see Figure 4. For the nine tests with $\Delta\sigma/\sigma_y < 1$, the life following an overload was in each case longer than the average life with no overload, that is, $N_{ov}/\bar{N} > 1$. For the six tests with $\Delta\sigma/\sigma_y > 1$, all but one gave $N_{ov}/\bar{N} < 1$. In order to define the two types of effect on fatigue life and the boundary between them, least squares fits of the two sets of data were performed. Since the data with $\Delta\sigma/\sigma_y < 1$ showed a smaller range in $\Delta\sigma/\sigma_y$ than in $N_{ov}/\bar{N}$, the variation in $\Delta\sigma/\sigma y$ was minimized to obtain the least squares line. The other data group showed a smaller range in $N_{ov}/\bar{N}$, so this parameter was minimized. The resulting least squares lines confirm and quantify the apparent trend of Figure 4, that tensile overload increases notch fatigue life for loading in which $\Delta\sigma < \sigma_y$. For fatigue loading in which $\Delta\sigma > \sigma_y$, overload decreases life.

Thermal Overload

Tests have been performed using specimens of the type in Figure 1(a), to determine if thermal stresses can be used to produce beneficial overloads. In the same general way that a mechanical overload produces compressive residual stress at a notch, thermal loading can be used, in principle, to produce favorable residual stress at a notch. The tests were performed to determine if the principle could be put to practice.

The test conditions for the thermal overload tests are outlined in Table II. Three combinations of specimen geometry and cyclic loading were used which were identical to three of the conditions for the baseline fatigue life and mechanical overload tests described in Table I.

TABLE II. CONDITIONS FOR THERMAL OVERLOAD TESTS

Specimen	ρ mm	F_{max} kN	Thermal Loading Location	Coolant	ΔT	N Cycles	$\bar{N}$ Cycles
#1	1.3	190	notch interior	air	near zero	4,200	4,330
#2	1.3	190	notch interior	water	-	4,800	4,330
#3	1.3	76	notch interior	water	-	90,600	76,000
#4	6.4	15.6	outer diameter	water	111°C	60,000	4[illegible],600
#5	6.4	15.6	outer diameter	water	224°C	172,000	44,600

The thermal overload procedure is shown in Figure 5. The test specimens were made from the same two cylinders as the earlier mechanical overload tests. These cylinders had overstrain residual stresses present before being cut into specimens. The overstrain residual stresses were reintroduced into each of the thermal overload specimens using a clamp as shown in Figure 5. Resistance strain gages, applied to the inner diameter surface at the notch location, were used to determine the clamping conditions for the tests. The specimens were clamped so that the circumferential direction strain reading was returned to that measured from the specimen with the notch present, but before the cut through the wall was made. Thus, the thermal loading was performed on specimens with an outer diameter surface tensile stress already applied which was about the same magnitude as the residual tensile stress present in the overstrained cylinders. The circumferential direction tensile residual stress at the outer diameter surface of the cylinders considered here, that is, for nominal outer-to-inner diameter ratio of 2.0 and material yield strength of 1200 MPa, was about 700 MPa (ref 3).

The actual thermal overload occurred when the specimen and clamp assembly was removed from a furnace at 370°C and coolant was immediately applied. Specimen #1 was a control with no coolant applied except for natural convection air. No significant temperature gradient nor associated effect on fatigue life was expected or observed. Specimens #2 and #3 were cooled with water sprayed only into the notch. All other surfaces of the specimen were protected from water contact. Relatively small and possibly significant increases in fatigue life were observed in these specimens with notch cooling.

The most significant thermal overload tests were with specimens #4 and #5. Before testing, thermocouples were welded on the inner and outer diameter surfaces at an angular location of 45° from the notch. The specimen faces and inner diameter surface were covered with glass fiber insulation so that only the outer diameter surface would be subject to coolant. The specimens were heated to 370°C and then cooled by immersion in room temperature water. Temperature recordings, taken every three seconds, showed that the maximum temperature difference, ΔT, between inner and outer surfaces occurred after about twelve seconds. The ΔT values are shown in Table II.

A comparison of fatigue life results from thermally overloaded specimens with the previous results from mechanically overloaded specimens is shown in Figure 6. Although only five thermal tests were performed, encouraging trends can be identified. First, no significant decrease in fatigue life has thus far been observed for specimens subjected to thermal overload. Second, a significant increase in fatigue life can be produced by thermal overload, as indicated by the nearly fourfold increase in life of specimen #5 compared to identical specimens with no overload.

SUMMARY

Both the stress concentration factor approach, $\sigma_{nom} K_t$, and the fracture mechanics approach, $K/\rho^{1/2}$, for calculating stress range give a good representation of notch fatigue life. This indicates that both approaches provide adequate measures of notch root stress during fatigue loading for a range of material properties and notch geometries.

The point at which stress range at the notch root, $\Delta\sigma$ equals yield strength, σ_y, is a clear boundary between two overload effects, above which overload reduces notch fatigue life and below which overload increases life, by a factor of 10 or more in some cases. This indicates that fatigue loading with $\Delta\sigma > \sigma_y$ overwhelms and relieves the compressive residual stress produced by overload, so that the remaining effect of overload is to use up some of the notch toughness of the material and result in a decrease in fatigue life.

Results thus far indicate that a thermal overload can also produce, for appropriate conditions, an increase in fatigue life, possibly as significant an increase as that due to a mechanical overload. Further, no significant decrease in fatigue life has been observed for thermally overloaded specimens, even when fatigue tested under conditions for which mechanical overload causes a decrease in life. Thus, it appears that thermal overload does not cause the same damage to material toughness as that which can be caused by mechanical overload.

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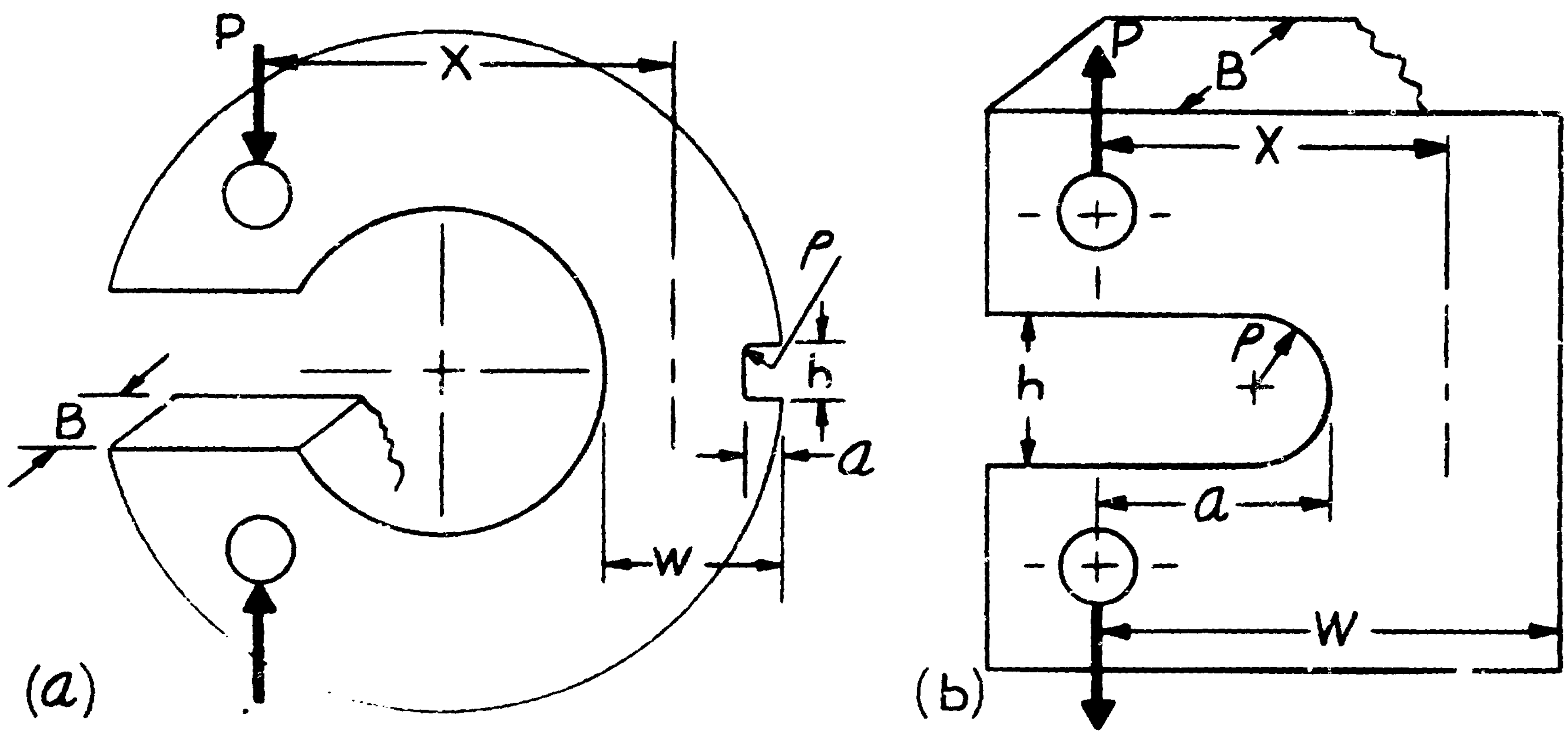

Figure 1. Two types or notch test specimen: (a) compression loaded disk; (b) tension loaded compact type.

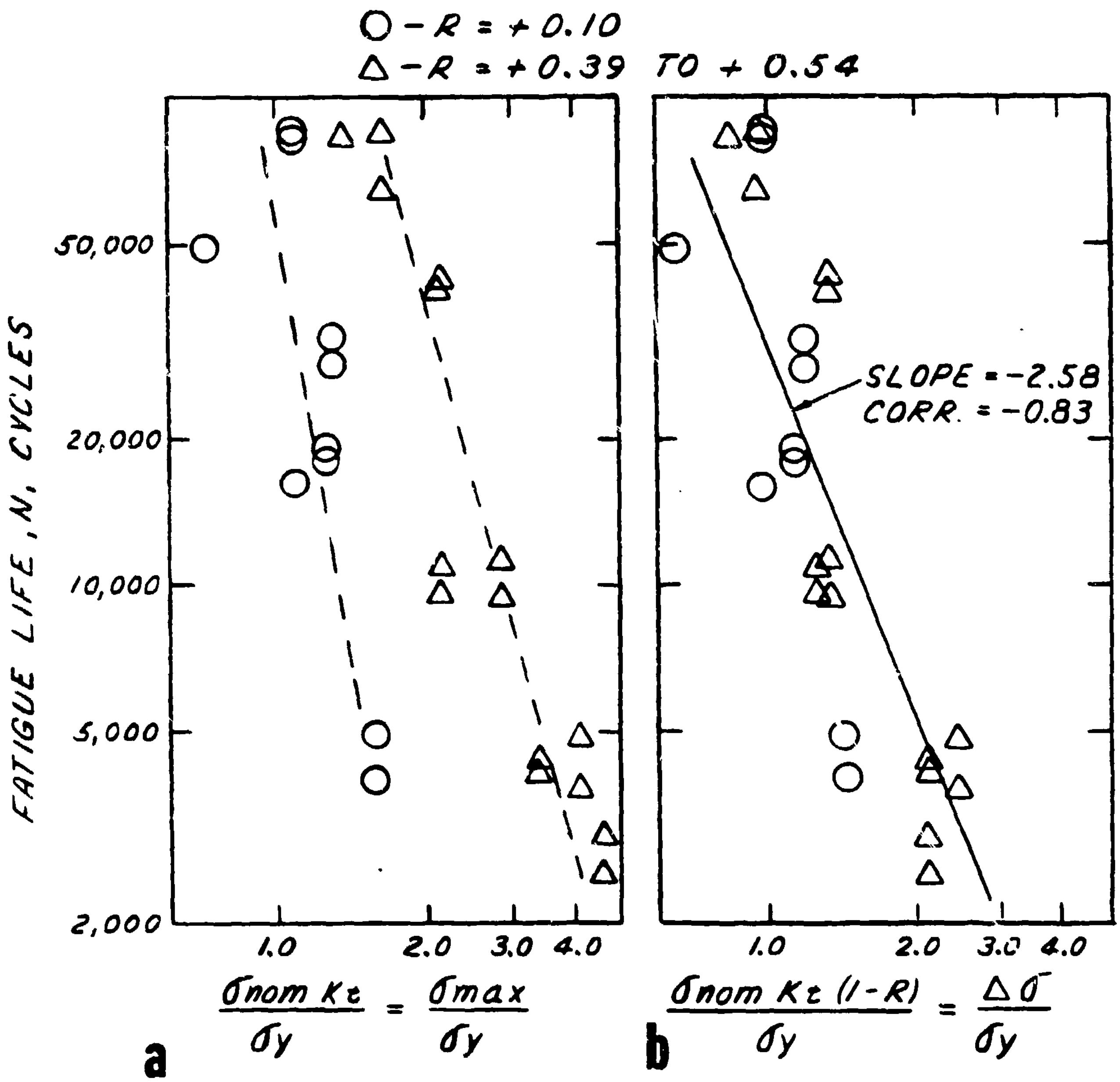

Figure 2. Fatigue life versus σ_{max} and $\Delta\sigma$ calculated using K_t approach.

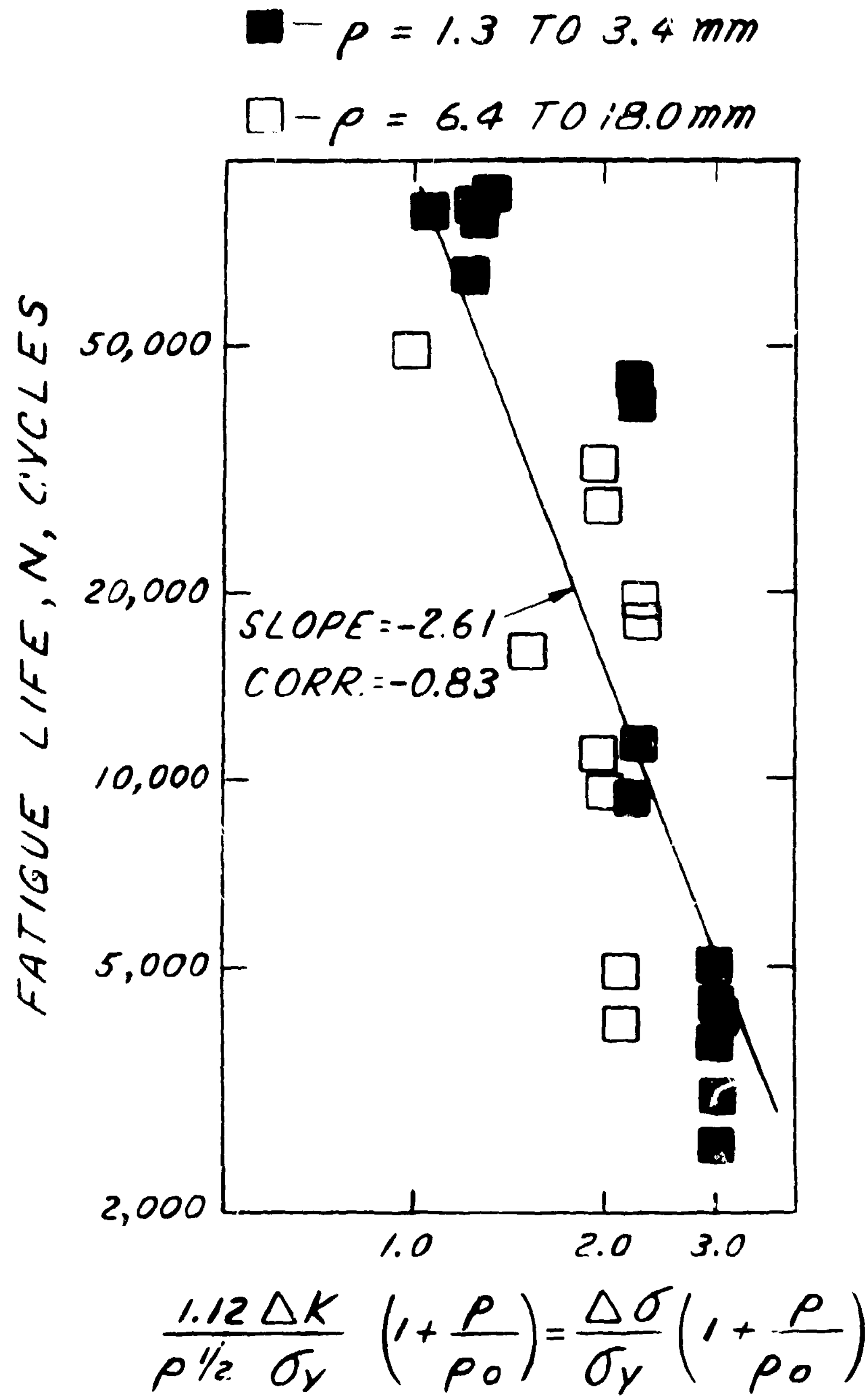

Figure 3. Fatigue life versus $\Delta\sigma$ calculated using $K/\rho^{1/2}$ approach.

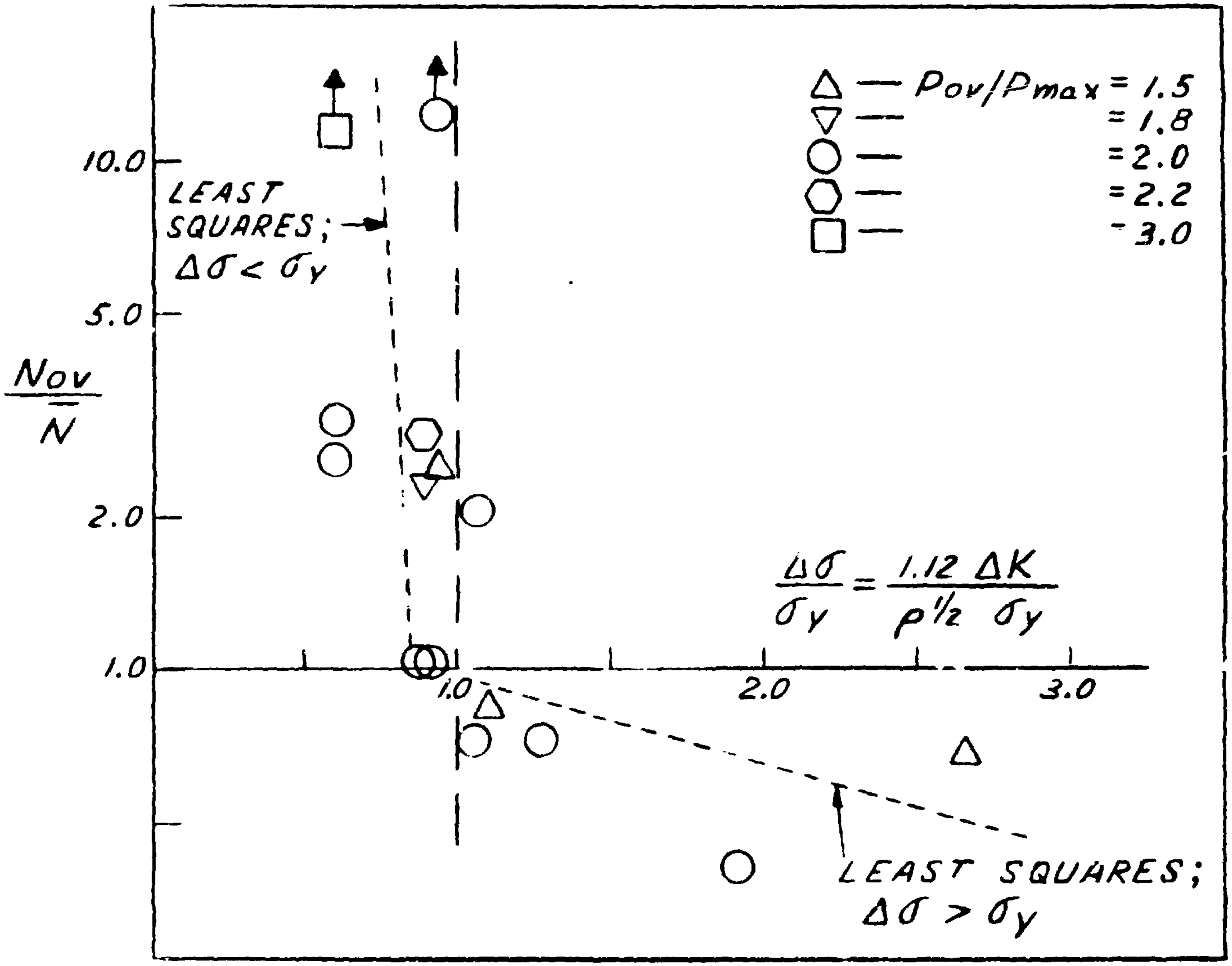

Figure 4. Ratio of fatigue life following an overload to average fatigue life with no overload versus $\Delta\sigma$ relative to yield strength, σ_y; for mechanical overload.

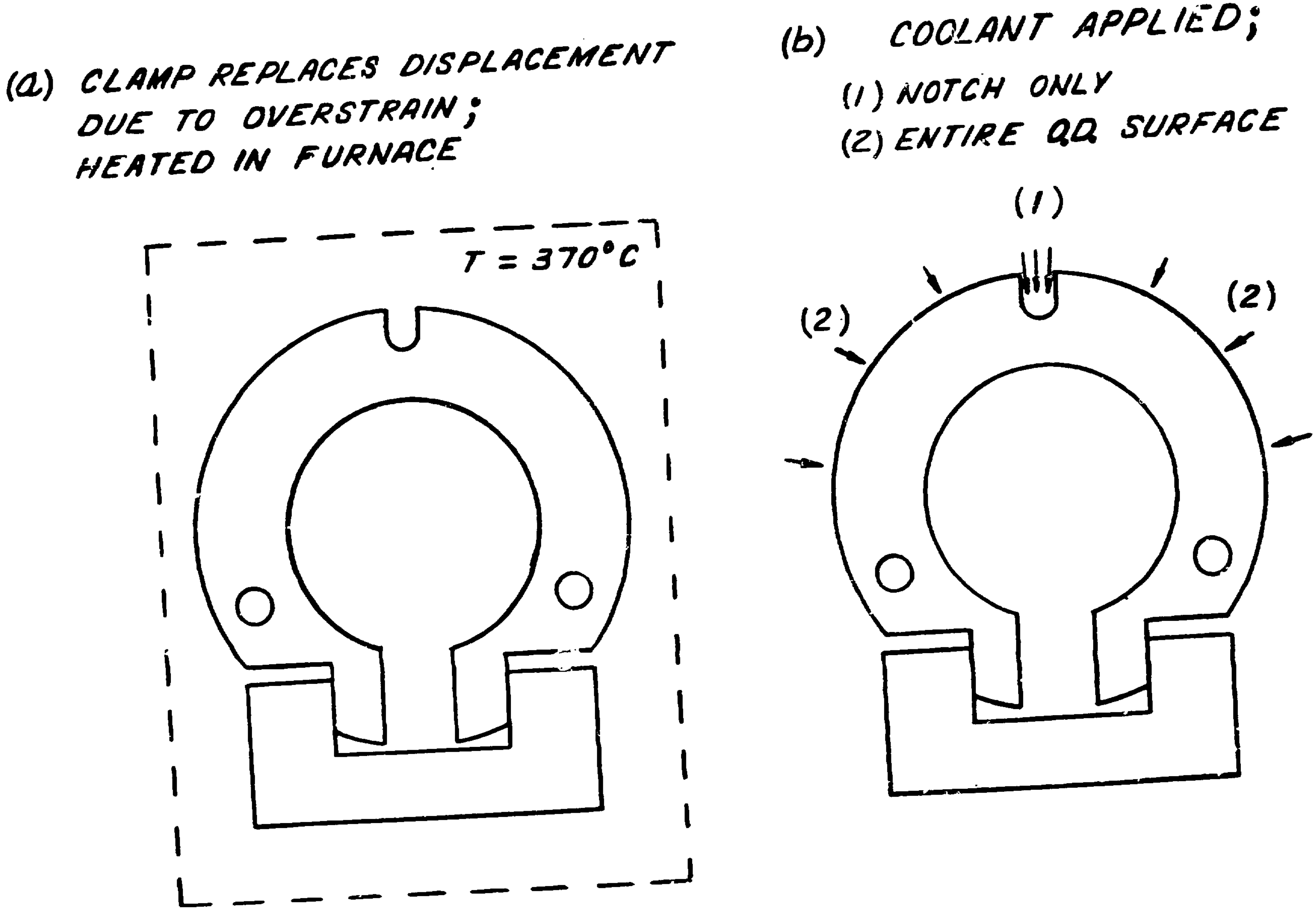

Figure 5. Sketch of thermal overload procedure.

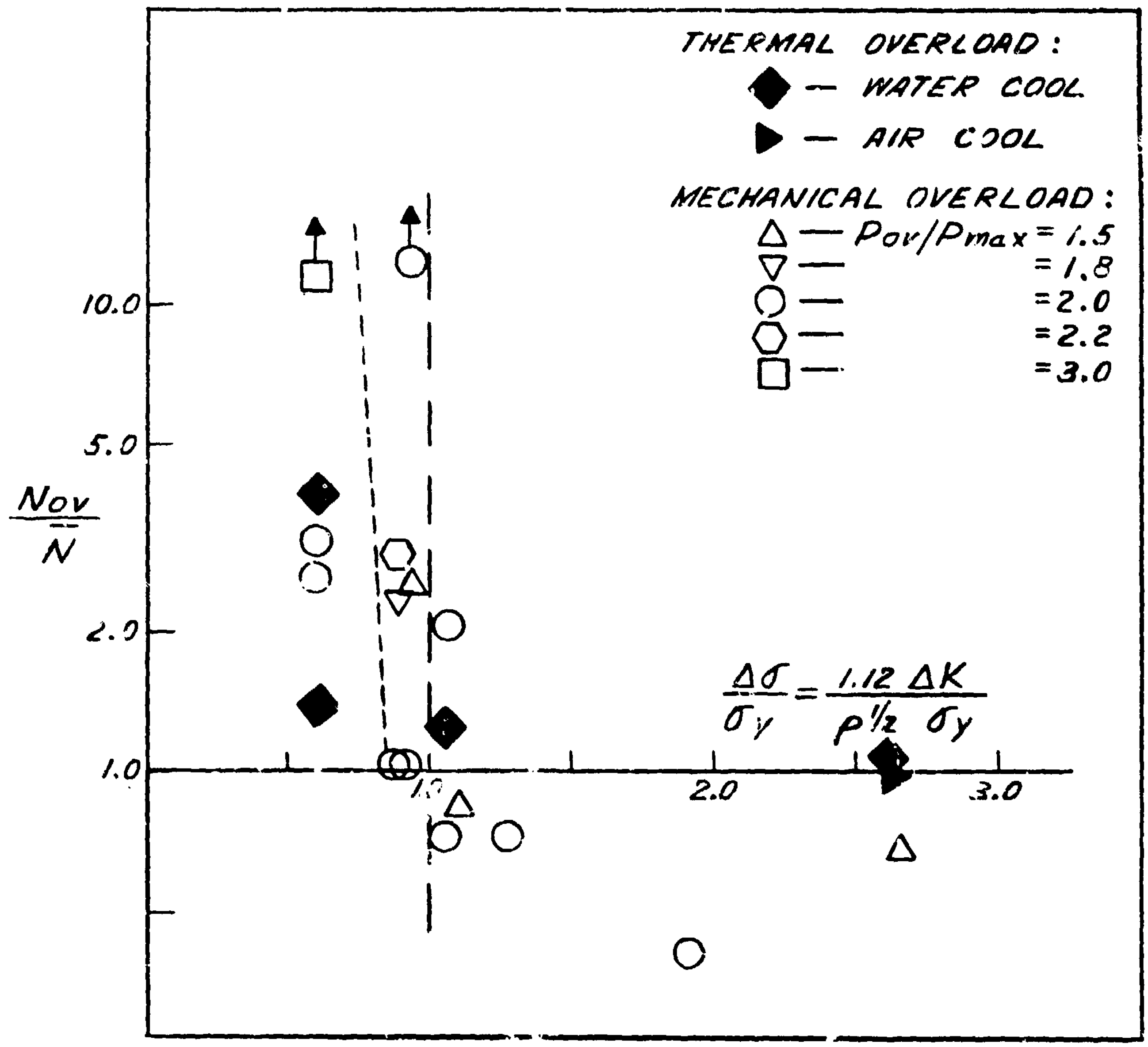

Figure 6. Ratio of fatigue life following an overload to average fatigue life with no overload versus $\Delta\sigma$ relative to yield strength, σ_y; for mechanical and thermal overloads.

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